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Methodology for assessing the dynamic properties of transmissions forestry machines with a bar working body

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Abstract. The studies talk about the features of moldboard and milling tillage. An analysis of current trends in the field of soil cultivation allows us to conclude that currently special attention is paid to reducing the energy intensity of the processing process. Forestry machines with bar working bodies have undoubted advantages over moldboard and milling machines, since during their operation they require much less power. Two transmission schemes were considered - chain and branched. When considering chain circuits with several variants of flywheels and passive working bodies, significant amplitudes of oscillations in transmissions appeared in the region of natural frequencies, and resonance phenomena when the frequencies of forced oscillations coincided with natural ones. Calculations have shown that an increase in the moment of inertia of the tractor engine flywheel by 3.14 times does not have a significant effect on the change in the natural frequencies of oscillations in the propulsion drive transmission, but causes them to decrease by 1.4 times. It follows that when the moments of inertia of rotating masses change, it is necessary to analyze the vibrations that occur in branched transmissions with active bar working bodies with significant moments of inertia.

1 Introduction

One way to determine the rational degree of load for the engine during furrow plowing in forestry operations is to conduct a thorough analysis of the dynamic response of the unit to overloads. This involves studying the engine's external characteristics under varying load conditions, as well as measuring the resistance forces on the working element. By understanding how the engine responds to different levels of load and how these loads affect

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the performance of the unit, operators can find the optimal balance of load that allows for efficient and effective furrow plowing without causing undue stress or strain on the engine. Additionally, proper maintenance and monitoring of the engine during forestry operations is essential to ensure that it remains in good working condition and can handle the fluctuations in load that occur during furrow plowing. Regular inspections, lubrication, and adjustments can help prolong the life of the engine and prevent breakdowns or malfunctions due to overloading [1, 2, 3].

When the unit operates at optimal speed, the value of the dynamic response increases and, due to the more complete use of kinetic energy to overcome overload areas, its productivity increases.

When creating forestry units for this purpose, it is possible to change, in accordance with calculations, the values of engine power, the moments of inertia of the flywheel or bar operating element, as well as the speed of movement. Increasing the power of the tractor engine will always ensure an increase in the productivity of the unit, but the value of its load factor K_l will remain low. It is necessary to achieve by calculation the optimal combination of all variable parameters of the units, which will ensure an increase in their productivity, efficiency and a reduction in fuel consumption for the amount of work.

Additionally, it is important to consider the damping characteristics of the system in order to avoid excessive vibrations and ensure smooth operation. Damping can help dissipate the energy of the vibrations and prevent them from building up and causing damage.

Overall, the use of mathematical models and analysis techniques can greatly help in optimizing the design and operation of forestry units, ensuring their reliability and efficiency. By considering the dynamic parameters and potential resonant frequencies, engineers can make informed decisions to mitigate the risks of vibration-related issues in transmission systems.

The purpose of the study is to investigate the energy and dynamic parameters of the bar working body of a forestry machine.

The study aims to analyze and improve the performance of the bar working body of forestry machines during wood cutting operations. The indicators that determine the efficiency of the working body will be identified and studied in depth.

The proposed algorithm based on the Runge-Kutta method in the MathCAD system will help in solving the system of differential equations that govern the behavior of the bar working body. The eigenvals functions will aid in finding the transfer functions of the dynamic system, allowing for a comprehensive analysis of its performance.

By calculating the amplitude-frequency characteristics of the dynamic systems involved in wood cutting operations, valuable insights can be gained into the behavior and efficiency of the bar working body. This information can then be used to make improvements and optimizations to enhance the overall performance of forestry machines in wood cutting tasks.

The task is to obtain areas of change in factors leading to an optimal solution to this problem. For a tractor unit operating on strip preparation of soil for planting trees, the speed of movement should be determined from the premise that the duration of the overload, which exceeds the torque developed by the tractor engine, must correspond to the optimal time for the engine flywheel to release the accumulated kinetic energy. To do this, it is necessary to determine the operating speed of the unit that meets this requirement [4, 5, 6].

2 Materials and methods

The objects of research are forestry and forest reclamation tractors, which differ in their design, engine power and dynamic parameters, propulsion designs and layout. When the above-mentioned units operate, the short-circuit load factor K_l of their engines should be close to 0.9. However, when performing basic energy-intensive work in forestry, K_l remains significantly lower, which indicates irrational use of energy due to the discrepancy between the dynamics of external conditions and the dynamic characteristics of the units [7–9].

In agriculture, when working on plowing, periods of load fluctuations on the crankshaft of tractor engines last 0.2–0.5 s; in the forestry industry, when hauling timber, it is 0.8–1.0 s. At the same time, in forestry, when performing basic energy-intensive work related to soil preparation and strip clearing of low-value plantings, the duration of load fluctuations is 3–5 s. This leads to a transition of engine operation to a non-regulatory branch of the external characteristic, a reduction in the engine crankshaft speed below the permissible ones and, as a consequence, the need to carry out the technological process at low (0.5–0.6) values of its load factor [9].

Analysis of dynamic models of forestry machines based on tractors showed that the most common are mathematical models of complex torsional oscillatory systems [10, 11].

The algorithm involved first defining the equations of motion for the system based on the moments of inertia of the engine flywheel and the cutting bar, as well as the damping coefficients and stiffness of the system. This resulted in a set of second-order differential equations that described the motion of the system.

Next, the Runge-Kutta method was used to numerically solve the differential equations and find the natural frequencies and modes of the system. The "eigenvals" function in MathCAD was then used to calculate the transfer functions of the system, which provided insight into how the system responds to different input frequencies.

Finally, the AFCs were plotted to visualize how the amplitude of the system's oscillations varied with frequency. This information was crucial for understanding the dynamic behavior of the peat cutting machine and optimizing its design for stability and performance.

Overall, the algorithm and program developed for calculating free oscillations of the peat cutting machine provided valuable insights into the system's behavior and helped engineers make informed decisions about its design and operation [12].

Frequency characteristics for chain and branched transmission schemes when using implements driven by a power take-off shaft (PTO) were determined to evaluate the dynamic properties of transmissions of tractor units. The conclusions about changes in loads depending on changes in gear ratios, observed from the dynamic characteristics, are also confirmed when calculating loads as a result of forced vibrations of rotating transmission parts from disturbing influences.

Modernization of the unit by changing the moment of inertia of the rotating masses of the unit should be carried out primarily in the active bar working bodies by changing the moments of inertia, and the engine flywheel cannot be changed, since this requires modernization of the tractor.

During the operation of a forestry unit, the kinetic energy of the rotating masses is accumulated in its dynamic system, which helps to overcome overloads. The values of the possible return of kinetic energy are determined by the magnitude of the dynamic response of the system to forced overloads (unit dynamics) $K_{av.opt.}$, which determines the ability of the tractor engine, together with the inertial masses of the dynamic system of the unit attached to its shaft, to overcome overload areas due to an increase in torque from the action of inertial forces taking into account the adaptability coefficient (current value), K'_{ad}.

$$K_{opt.} = \frac{M_r^{\max}}{M_e} \text{ or } K_{opt.} = K'_{ad} + \frac{J_\Sigma \xi_{n,n-1}}{M_e}, \tag{1}$$

where $M_r^{\max}$ – moment of resistance at the overload section (maximum value); J_Σ – moment of inertia reduced to the engine crankshaft; $\zeta_{n,\ n-1}$ – angular acceleration of the crankshaft during the nth second of overload action; M_e - torque developed by the tractor engine.

Calculation of $K_{opt.}$ can be executed using special programs on a PC. The influence of resistance forces on the working bodies from the heterogeneity of the developed medium can be characterized by the dynamic coefficient K_d and duration τ of the area of increased load. The registration of these values was carried out by the author while the tractor unit was operating under real production conditions, recording the torque values on the engine shaft and the experiment time on oscillograms. For overload areas, correlation dependences of $K_d=f(\tau)$ or $\tau=f(K_d)$. The concept of the coefficient of correspondence of the dynamic parameters of the unit to the dynamics of external conditions was introduced into the characteristics of the unit $K_c=K_{opt.}/K_d$, which for the majority of forestry and forest reclamation units does not exceed 0.7.

Figure 1 shows graphs of the dependences $K_c=f(\tau)$ for operating conditions of varying severity. It follows from the graphs that with increasing rigidity of the dynamic component and the duration of overload sections, the dependence curve $K_c=f(\tau)$ tends to decrease. The dynamic response and K_c increase with an increase in the reduced moment of inertia, the adaptability coefficient, and the difference in the angular velocity of the crankshaft and decrease with an increase in engine power.

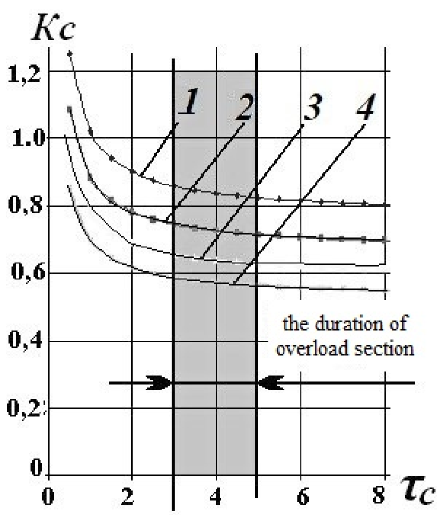

Fig. 1. Dependence $K_c=f(\tau)$: 1 – K_d=1,3; 2 – K_d=1,5; 3 – K_d=1,7; 4 – K_d=1,9.

Optimization of the energy and dynamic parameters of the unit can be performed based on the equations:

$$K_{l.opt.} = 1-(K_d - K_{opt.}), \quad (2)$$

$$V_{opt.}t_{opt.} = V_{d.}t_{d.}, \quad (3)$$

$$N_{e.opt.} = \frac{N_{r.v.}V_{opt.}}{K_{l.opt.}V_{r.v.}}K_u, \quad (4)$$

$$J_{\Sigma opt.} = \frac{[K_d - (1-K_{l.opt.}) - K_n^{'}]M_e t_{opt.}}{\Delta\omega_{opt.}}, \quad (5)$$

where $K_{l.opt.}$, $V_{opt.}$, $V_{d.}$, $V_{opt.}$, $N_{e.opt.}$, $\Delta_{\omega opt.}$ – optimal values of K_l, movement speeds, engine power, reduction of the angular speed of the crankshaft; $N_{r.v.}$, $V_{r.v.}$ – real (from experience) values of power consumption and movement speed.

By optimizing the energy and dynamic parameters of the unit using expressions (2–5), for a specific unit operating under an unsteady load, it can be determined that the calculated values of the optimal moment of inertia of the rotating masses $J_{\Sigma opt.}$ exceed the existing ones by 83%. Increasing the performance indicators of tractor units should consist of simultaneous selection of their energy, dynamic and speed parameters.

By analyzing the dynamic behavior of the transmission system, engineers can determine the critical speeds at which resonant oscillations may occur. This allows them to make necessary adjustments to the design of the transmission unit to prevent potential failures. Additionally, by optimizing the dynamic load of the power transmission system, the overall efficiency and performance of the tractor unit can be improved.

Computer simulations and modeling techniques are often utilized to predict the dynamic behavior of the transmission system under varying operating conditions. This enables engineers to identify potential issues and make informed decisions to optimize the design and performance of the transmission unit.

Overall, the study and optimization of the dynamic load of power transmissions in tractor units are crucial for ensuring the reliability and efficiency of the machinery. By utilizing advanced modeling and simulation techniques, engineers can effectively analyze and optimize the performance of the transmission system to meet the demands of modern agricultural machinery. For units such as a bar actuator driven by a tractor PTO, it is necessary to consider branched ten-mass systems (Figure 2 a, b).

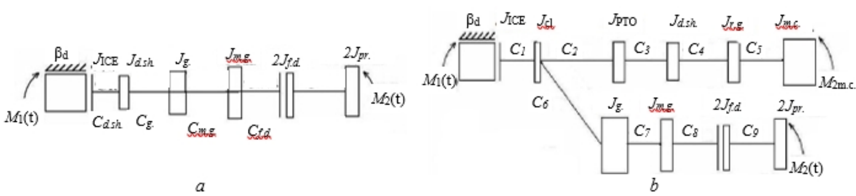

Fig. 2. Equivalent dynamic circuit: *a* – Six-mass equivalent dynamic circuit, *b* – Branched ten-mass equivalent dynamic diagram of the system when using the main power flow through the PTO of a tractor with a classic kinematic diagram of the DT–75B type; *J* – reduced moments of inertia of tractor and implement mechanisms; *C* – given stiffness coefficients of transmission sections; $M_1(t)$ – torque from engine; $M_2(t)$ – moment of resistance from the medium being developed.

The upper branch of the working body drive transmission according to the diagram, including the rotating masses of the engine with a flywheel J_{ICE} and clutch J_{cl}, a power take-off shaft with a gearbox J_{PTO}, a driveshaft of the working body drive $J_{d.sh.}$ with reduction gear $J_{r.g.}$, cutter drive shaft with cutter assembly $J_{m.c.}$.

The lower branch is the drive of the tractor chassis from the gearbox J_g, through the main gear $J_{m.g.}$, final drives $J_{f.d.}$ and propulsors J_{pr}.

By introducing the notation for the differential equation

$$a_k = -\left(b_k \cdot s + c_k\right),\ d_k = J_k \cdot s^2 + \left(b_{k-1} + b_k\right) \cdot s + c_{k-1} + c_k,$$

let's find the determinant $\mathrm{D_{kn}}$ of a tridiagonal matrix using the recurrent formula

$$D_{kn} = d_k \cdot D_{k+1\,n} - a^2{}_{k+1} \cdot D_{k+2\,n},\ D_{k\,k} = d_k,\ k = 1, \dots, n. \tag{6}$$

The determinants for the above branched systems are not tridiagonal, but they are expressed through tridiagonal determinants of lower order, respectively, according to the formulas:

$$1.\ \Delta = D_{1n}; \tag{7}$$

$$\Delta = D_{1\,k+m} \cdot D_{k+m+1\,n} - a^2{}_{k+m+1} \cdot D_{1\,k-1} \cdot D_{k+1\,k+m} \cdot$$
$$D_{k+m+2\,n} - a^2{}_{k+m} \cdot D_{1\,k+m-1} \cdot D_{k+m+1\,k+2m} \cdot$$
$$2.\ \cdot D_{k+2m+2\,n} + a^2{}_{k+m} \cdot a^2{}_{k+m+1} \cdot D_{1\,k\_1} \cdot D_{k+1\,k+m} \cdot \tag{8}$$
$$D_{k+m+2\,k+2m} \cdot D_{k+2m+2\,n} - 2 \cdot D_{1\,k-1} \cdot a_k \cdot a_{k+1} \cdot$$
$$\cdots\cdots a_{k+2m+1} \cdot D_{k+2m+2\,n};$$

$$\text{a) } \Delta = D_{1\,k+m} \cdot D_{k+m+1\,k+2m} - a^2{}_{k+m} \cdot D_{1\,k-1} \cdot D_{k+1\,k+m} \cdot D_{k+m+2\,k+2m}; \tag{9}$$

$$\text{b) } \Delta = D_{1\,k+m} \cdot D_{k+m+1\,k+m+1} - a^2{}_{k+m} \cdot D_{1\,k-1} \cdot D_{k+1\,k+m} \cdot D_{k+m+2\,k+m+1}. \tag{10}$$

Based on the above, we will calculate the transfer functions for torques on transmission shafts in the form of a chain n+1 mass oscillatory dynamic system. The equations of resonant oscillations have the form:

$$\begin{array}{l} J_1 \cdot \ddot{y}_1 + b_1 \cdot (\dot{y}_1 - \dot{y}_2) + c_1 \cdot (y_1 - y_2) = P_1(t) \\ \text{------------------------------} \\ J_k \cdot \ddot{y}_k - b_{k-1}(\dot{y}_{k-1} - \dot{y}_k) - c(y_{k-1} - y_k) + b_k(\dot{y}_k - \dot{y}_{k+1}) + c_k(y_k - y_{k+1}) = P_k(t) \\ \text{--} \\ J_{n+1}\ddot{y}_{n+1} - b_n(\dot{y}_n - \dot{y}_{n+1}) - c_n(y_n - y_{n+1}) = P_{n+1}(t) \end{array} \tag{11}$$

Let us divide the k-th equation by J_k, subtract each subsequent equation from the previous one and, introducing into it the variables $\varphi_k = y_k - y_{k+1}$ – the angle of twist of the k-th shaft, we obtain a system of differential equations for shafts of the nth order

$$\begin{array}{l} d_1(s)\phi_1 + a_2(s)\phi_2 = P_1(s)/J_1 - P_2(s)/J_2 \\ \text{------------------------} \\ q_k(s)\phi_{k-1} + d_k(s)\phi_k + a_{k+1}(s)\phi_{k+1} = P_k(s)/J_k - P_{k+1}(s)/J_{k+1} \\ \text{------------------------------------} \\ q_n(s)\phi_{n-1} + d_n(s)\phi_n = P_n(s)/J_n - P_{n+1}(s)/J_{n+1}, \end{array} \tag{12}$$

where

$$\begin{array}{l} d_k(s) = s^2 + b_k(1/J_k + 1/J_{k+1})s + \\ + c_k(1/J_k + 1/J_{k+1}), \\ a_k(s) = -(b_k + c_k)/J_k, \\ q_k(s) = -(b_{k-1}s + c_{k-1})/J_k, \quad s \equiv d/dt. \end{array} \tag{13}$$

Applying the Laplace transform to equations in which zero initial conditions are assumed, we ultimately obtain a system of algebraic equations in matrix form, resolved with respect to images of variables $\varphi_k(s)$

$$\Phi(s) = W(s) \cdot P(s), \tag{14}$$

where the matrix of transfer functions has the form

$$W(s)=\frac{1}{D_{1n}}\begin{bmatrix} D_{2n} & -a_2 D_{3n} & \cdots & (-1)^{1+n} a_2..a_n \\ -q_2 D_{3n} & D_{11} D_{3n} & \cdots & (-1)^{2+n} D_{11} a_3..a_n \\ q_2 q_3 D_{4n} & -D_{11} q_3 D_{4n} & \cdots & (-1)^{3+n} D_{12} a_4..a_n \\ \cdots & \cdots & \cdots & \cdots \\ (-1)^{k+1} q_2..q_k D_{k+1n} & (-1)^{k+2} D_{11} q_3..q_k D_{k+1n} & \cdots & (-1)^{k+n} D_{1k-1} a_{k+1}..a_n \\ \cdots & \cdots & \cdots & \cdots \\ (-1)^{n+1} q_2..q_n & (-1)^{n+2} D_{11} q_3..q_n & \cdots & D_{1n-1} \end{bmatrix} \tag{15}$$

and

$$D_{kn}=\begin{Vmatrix} d_k(s) & a_{k+1}(s) & 0 & \cdots & 0 \\ q_{k+1}(s) & d_{k+1}(s) & a_{k+2}(s) & \cdots & 0 \\ 0 & q_{k+2}(s) & d_{k+2}(s) & \cdots & 0 \\ \cdots & \cdots & \cdots & \cdots & \cdots \\ 0 & 0 & 0 & \cdots & d_n(s) \end{Vmatrix} \tag{16}$$

determinant of a tridiagonal matrix calculated by recurrence relation

$$D_{k\,n}=d_k(s)\,D_{k+1\,n}-a_{k+1}(s)\,q_{k+1}(s)\,D_{k+2\,n},\quad D_{k\,k}=d_k(s). \tag{17}$$

As a result, we arrive at similar, but slightly modified formulas, since for the n-shale system under consideration we obtain a similar asymmetric matrix. The value of the torque on the k – ohm shaft is determined by the expression

$$M_k(s)=c_k\varphi_k(s),$$

then the transfer function, which expresses the relationship between the torque on the *k*-th transmission shaft and the disturbing force $P_l(t)$ applied to the first mass, is presented as:

a) for $l \le k$

$$W_{1\,k}(s)=\frac{(-1)^{k+1}\, c_k q_{1+1} q_{1+2}\cdots q_k D_{1\,1-1}\left(D_{k+2\,n} q_{k+1}+D_{k+1\,n}\right)}{J_1 D_{1n}}, \tag{18}$$

b) for $l = 1$

$$W_{1\,k}(s)=\frac{(-1)^{k+1}\, c_k q_2 q_3\cdots q_k D_{k+1\,n}}{J_1 D_{1n}}, \tag{19}$$

c) for $l > k$

$$W_{1\,k}(s)=\frac{(-1)^{k+1}\, c_k D_{1\,k-1} a_{k+1} a_{k+2}\cdots a_{1-1}\left(D_{1\,n}+a_1 D_{1+1n}\right)}{J_1 D_{1n}}, \tag{20}$$

d) for $l = n+1$

$$W_{n+1\,k}(s)=\frac{(-1)^{k+n+1}\, c_k D_{1\,k-1} a_{k+1} a_{k+2}\cdots a_n D_{k+1\,n}}{J_{n+1} D_{1n}}. \tag{21}$$

Based on the presented method for a ten-mass model of a forestry unit, the transfer functions of disturbances from the tractor engine, its caterpillar engagement and soil resistance to an arbitrarily selected *k*-th transmission shaft, respectively, have the form

$$W_{ICE\,k}\left(s\right)=\frac{\left(-1\right)^{k+1}\,c_k q_1 q_2\cdots q_k D_{k+1\,9}}{J_1 D_{1\,9}},$$
$$W_k\left(s\right)=\frac{\left(-1\right)^{k+8}\,c_k D_{1\,k-1}a_{k+1}a_{k+2}\cdots a_7\left(D_{8\,9}+a_8 D_{9\,9}\right)}{J_8 D_{1\,9}}, \tag{22}$$
$$W_{c\,k}\left(s\right)=\frac{\left(-1\right)^{k+9}\,c_k D_{1\,k-1}a_{k+1}a_{k+2}\cdots a_9}{J_{10} D_{1\,9}}.$$

Crawler tractors differ from wheeled tractors in that the driver has a significant impact on the transmission. The manifestation of disturbances from the track engagement is observed on the clutch shaft of the unit. Frequency ranges from the influence of caterpillar engagement are determined by the formula

$$\omega_{cat}=2\cdot\pi\cdot n_k\cdot z_k=\frac{2\cdot\pi\cdot n_{ICE}\cdot z_k}{i_{tr}}=\frac{v\cdot z_k}{r_k}, \tag{23}$$

where n_{cr} and n_{tr} – rotation speed of the crankshaft of the tractor engine and the drive sprockets of its tracks; z_k– number of teeth of tractor caterpillar drive sprockets; i_{tr} – gear ratio of its transmission; v – tractor speed; r_k– track sprocket radius.

Frequency ranges from the interaction of active working bodies with the developed environment are determined by the formula:

$$\omega_{w.b.}=\frac{2\cdot\pi\cdot n_{w.b.}\cdot z_{kn}}{i_{PTO}\cdot i_{w.b.}}, \tag{24}$$

where $n_{w.b.}$ – working body rotation speed; z_{kn} – number of knives (cutters) on the working body; i_{PTO} – gear ratio of the PTO gearbox; $i_{w.b.}$ – gear ratio of the implement gearbox.

3 Results and discussion

When the speed of movement of the forestry unit LHT-4 designed by All-Russian Research Institute of Forestry and Forestry Mechanization changes from 0.63 to 1.19 m/s (movement in I–IV gears), the frequency of the oscillatory effect from the caterpillar engagement changes in the range from 37.05 to 70.00 rad/s, for the LHT-55 tractor, the values of oscillation frequencies from the tracks in working gears are 37.01–64.76 rad/s, for the LHT-100A they will be 31.19–54.58 rad/s. The transmission of the reclamation tractor DT-75 B with a bar actuator experiences a frequency of caterpillar impact of 6.82–9.34 rad/s in winter and 16.16–22.27 rad/s in summer. On the gun side, it is loaded through the PTO with frequencies of 716.22–994.75 and 733.23–1018 rad/s, respectively.

The algorithm involves solving the system of differential equations that describe the free oscillations of the system. This includes taking into account the variable moments of inertia of the engine flywheel and the active bar working body. The Runge-Kutta method is used to numerically solve these equations.

The MathCAD system is utilized for this purpose, with the "eigenvals" function being used to calculate the eigenvalues of the system. These eigenvalues help in determining the natural frequencies of the system.

Using the transfer functions obtained from the eigenvalues, the amplitude-frequency characteristics (AFC) of the system are calculated. These characteristics provide important information about the system's behavior under free oscillations.

By carrying out these calculations, valuable insights into the dynamic behavior of forestry units can be gained. This information can be used to optimize the design and performance of such systems.

Frequency response analysis is a common method used to assess the dynamic properties of mechanical systems. In the case of transmissions in forestry and forest reclamation units, understanding the frequency response can provide valuable information about how the system will behave under different operating conditions.

In a chain and branched transmission scheme, the system consists of multiple interconnected components such as gears, chains, and shafts. Each component has its own natural frequencies and dynamic characteristics, which can impact the overall performance of the system. By analyzing the frequency response of the system, engineers can identify any potential issues such as resonance or vibration that may affect the system's operation.

When using implements with passive working bodies driven by a PTO, the frequency response of the system will also depend on the characteristics of the implement itself. For example, the mass distribution of the working bodies, the stiffness of the connection to the PTO, and the type of motion being performed all play a role in determining the frequency response of the system.

Overall, analyzing the frequency response of transmissions in forestry and forest reclamation units can help engineers optimize the design and operation of these systems to improve performance and reliability in the field.

In the chain transmission (Figure 3 a, b), the values of the moments of inertia of the tractor engine flywheel varied from 2.9 to 6 kg/m^2. The branched version compared options for changing the moments of inertia of the engine flywheel and the working element.

To address this issue, it is recommended to adjust the damping characteristics of the transmission system by modifying the stiffness and damping coefficients of the components involved. This can help to reduce the amplitude of oscillations and prevent resonances from occurring.

Additionally, implementing vibration isolators or dampeners at critical points in the transmission system can help to absorb energy and reduce the transmission of oscillations throughout the system. Proper maintenance and alignment of components can also help to minimize oscillations and improve overall performance.

Overall, addressing the frequency response and oscillation issues in the transmission system of the LHT-55 tractor with a PKL-70 plow requires careful analysis and potentially design modifications to enhance stability and reduce unwanted vibrations.

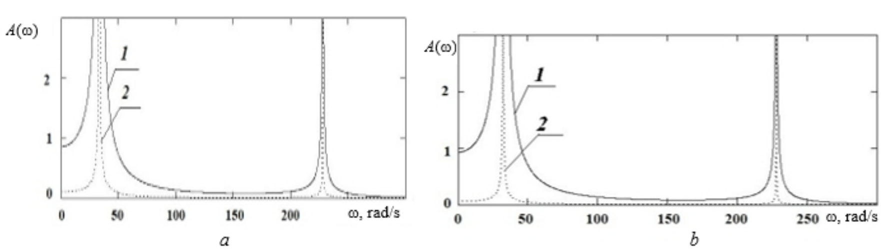

Fig. 3. The frequency response of the LHT-55 tractor transmission (1 gear): 1 – impact from the tracks on the gearbox shaft; 2 – impact from the engine to the final drive shafts; *a* – J_{max} = 2,9 kg·m^2, *b* – J_{max} = 6 kg·m^2.

Comparison of frequency response Figure 3 a, b shows that with an increase in J_{max} from 2.9 to 6 kg·m^2, no significant changes in transmission oscillation frequencies are observed. An increase in the moment of inertia of the engine flywheel does not cause resonance phenomena in the tractor transmission, but increases its performance by increasing movement speeds.

The frequency response graphs show that the changes in the moments of inertia of the engine flywheel and the working body have a significant impact on the dynamic behavior of the transmission. As the moment of inertia of the engine flywheel increases, the natural frequencies of the system shift towards higher values, leading to potential resonance issues. On the other hand, increasing the moment of inertia of the working body reduces the transmission's natural frequencies and helps to dampen oscillations.

It is crucial for designers and engineers to carefully consider the moments of inertia of the various components in the transmission system to ensure optimal performance and avoid potential resonance problems. Additionally, measures such as incorporating damping elements or adjusting the stiffness of the system can help mitigate any undesirable oscillations and ensure the smooth operation of the forestry unit.The operating speed of the unit during calculations was assumed to be 0.2 m/s according to the technical characteristics of the tools. The moment of inertia of the engine flywheel was assumed to be 2.9 and 8.0 kg·m^2, the moment of inertia of the working body reduced to the engine crankshaft was 1.15 and 3.61 kg·m^2.

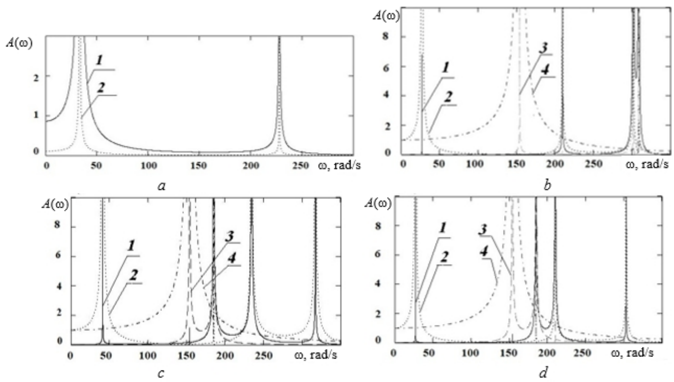

Fig. 4. Frequency response of the DT-75 transmission: 1 – impact from knives on the clutch shaft; 2 – impact from knives to the PTO; 3 – impact from the engine to the cutters; 4 – impact from the cutters to the gearbox; *a* – J_{ICE} = 2.9 kg·m^2; $J_{m.c.}$ = 1.15 kg·m^2; *b* – J_{ICE} = 2.9 kg·m^2; $J_{m.c.}$ = 3,61 kg·m^2; *c* – J_{ICE} = 8 kg·m^2; $J_{m.c.}$ = 1.15 kg·m^2; *d* – J_{ICE} = 8 kg·m^2; $J_{m.c.}$ = 3.61 kg·m^2.

The range of influence of tracks on the transmission is in the frequencies of 190–210 rad/s. And it increases with increasing moment of inertia of the engine flywheel.

From the analysis of the frequency response data of the DT-75 B tractor with a bar working element, it follows that the energy from the influence of the inertial mass of the cutters is concentrated within the frequencies of 40 and 230 rad/s. The energy of the impact of the caterpillar engagement on the transmission increases in the frequency range of 190–210 rad/s, the vibrations propagating through the transmission in the direction of the tractor gradually attenuate. From the analysis of Figure 4 a–d, we can draw preliminary conclusions that:

- increased vibrations are observed in the transmission section with minimal rigidity, that is, between the primary shaft of the creeper and the tractor PTO gearbox;
- an increase in the moment of inertia of the tractor engine flywheel does not cause significant changes in the frequency response and resonance phenomena in the transmission of the unit;
- an increase in the moment of inertia of the cutters in all cases reduces the level of vibrations in the transmission of the unit;
- frequencies from the influence of caterpillar engagement (6.8 and 22.3 rad/s) do not fall into resonance;

- frequencies caused by the interaction of knives with the developed external environment (peat deposit) 716–1018 rad/s do not have a significant effect on the dynamics of the unit transmission.

From the analysis of the amplitude-frequency characteristics of the dynamic model of the forestry unit based on the DT-75 tractor at different moments of inertia of the engine flywheel and blades, it can be noted that an increase in the moment of inertia of the engine flywheel has practically no effect on the change in the natural frequency of oscillations, an increase in the moment of inertia of the working body in 3.14 times causes a frequency shift towards decreasing them by 1.4 times.

An analysis of the transmission of a forestry unit based on a self-propelled chassis SSh-16 M with an end mill is presented in the frequency response graph in Figure 5. During the calculations, the number of knives on the cutter disk, movement and cutting speeds were varied. The speed of movement of the unit during testing was assumed to be constant at 0.4 m/s, the cutting speed was 8 and 16 m/s.

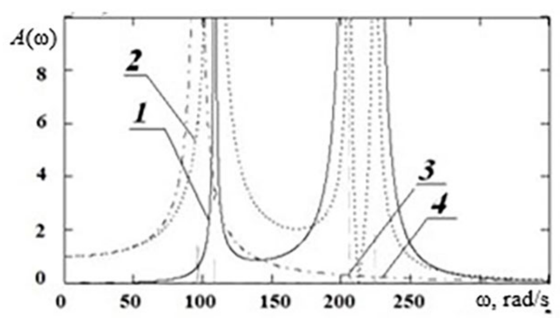

Fig. 5. Frequency response of the transmission of the self-propelled chassis SCH-16 with an end mill: 1 – impact from the cutter to the chassis clutch shaft; 2 – impact from the cutter to the PTO chassis; 3 – impact from the engine to the final drive shafts; 4 – impact from the movers to the clutch shaft.

The moment of inertia of the engine flywheel is 1.14 kg·m^2, the reduced moment of inertia of the bar working element is 0.16 kg·m^2. The transmission of a forestry unit based on a self-propelled chassis SCH-16 M is affected by the working element (end mill) depending on the adopted cutting speed of 8 or 16 m/s; the excited frequencies of 169.72 or 339.45 rad/s do not affect the nature of vibrations transmissions. Based on the frequency response of this unit, a preliminary conclusion can be made that its dynamic characteristics require scientific justification and design improvement in the direction of reducing the vibration amplitudes of the drive section of the milling working element.

The method mentioned likely refers to numerical modeling and simulation techniques in order to study the effects of gear ratios and transmission stiffness on the dynamic properties of a mechanical unit. By increasing the gear ratio of a tool with a bar working body, the rigidity of the transmission section decreases, leading to changes in the natural frequencies of the system.

It is important to calculate and analyze the natural frequencies of the system at zero damping in order to understand how the system will respond to external disturbances and vibrations. By detuning the natural frequencies from resonance, the system can potentially avoid harmful vibrations and maintain stability during operation.

Overall, the modeling and analysis of gear ratios and transmission stiffness are crucial in designing mechanical systems that are robust and efficient in terms of dynamic properties and characteristics. By optimizing these parameters, engineers can improve the performance and reliability of the system.

If necessary, damping devices can be installed in the transmission power circuit in the «PTO – working body» section. If we accept even slight damping (for example, with a damping decrement $\delta = 0.3$), then the amplitude values in the high-frequency region decrease.

When choosing the values of transmission gear ratios, it is necessary to carry out multi-criteria optimization according to the method. Modernization of the unit by changing the moment of inertia of its rotating masses should be carried out at the bar working parts; the engine flywheel must be changed only when modernizing the tractor.

4 Conclusion

By analyzing the mathematical models, engineers can also optimize the design of the transmission systems to improve efficiency and reduce wear and tear. Overall, the use of mathematical models in the design and modernization of agricultural machinery helps ensure that the machines operate smoothly, efficiently, and reliably. It allows engineers to make informed decisions and predict potential issues before they occur, ultimately leading to improved performance and longevity of the equipment. From the analysis of the frequency response of tractor units with plow implements, it can be noted that an increase in the moment of inertia of the engine flywheel from 2.9 to 6 kg·m^{2}does not cause significant changes in the oscillation frequencies of the tractor transmission and resonance phenomena in it. The frequencies from the impact of the track gear are not superimposed on the natural frequencies in the transmission.

Overall, the analysis highlights the importance of the design and material selection of components in bar working bodies with significant moments of inertia to minimize vibrations and ensure smooth operation of the transmission system in agricultural machinery. Additionally, increasing the moment of inertia of the bar working body can help reduce vibrations in the transmission system, while other factors like the engine flywheel and caterpillar gear do not significantly impact resonance frequencies. Further research and design optimization can help improve the performance and reliability of these units.

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